RESEARCH MEMORANDUM

DESIGN AND ENGINE EVALUATION OF A SEMISTRUT CORRUGATED

AIR-COOLED TURBINE BLADE FOR OPERATION AT

A TIP SPEED OF 1300 FEET PER SECOND

By André J. Meyer, Jr., Richard H. Kemp, and William C. Morgan

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Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

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UNCLASSIFIED

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SUMMARY

An improved air-cooled turbine blade is described, stress-analyzed, and evaluated in a full-scale turbojet engine. The construction consists of a sheet-metal outer shell and an internal strut of partial blade length. Corrugated sheet metal comprises the augmented cooling surface which is brazed between the strut and the outer shell. The effect on blade rupture life of varying outer-shell thickness, strut length, blade temperature distribution, and corrugation pitch and amplitude is presented graphically.

Blades fabricated in accordance with the design analysis were run at a turbine-inlet gas temperature of 1660° F and a tip speed of 1300 feet per second for 100 hours with 3.5-percent coolant flow. The same blades were then run for 25 hours with no coolant flow under the same engine conditions without failure.

INTRODUCTION

The desirability of increasing the turbine-inlet gas temperature has long been known. The strength limitations of even the best heat-resistant materials, however, make some form of added cooling for turbine blades very advantageous. Considerable research has been conducted on a wide variety of cooled turbine blade types. Most of this work has been summarized in references 1 and 2.

In the design of cooled blades several requirements must be fulfilled in order to obtain a good configuration. In approximate order of importance, these requirements for aircraft powerplants are: (1) adequate cooling with a minimum of coolant flow, (2) mechanical durability, (3) light weight, (4) a favorable chordwise and spanwise temperature
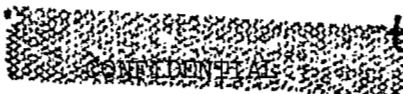
distribution in the blade, (5) ease of fabrication, and (6) low pressure drop through the coolant passages. The designer must devise the best arrangement of the blade components to obtain a good compromise among these requirements.

In general, convection-type air-cooled blades can be divided into three main categories: (1) Shell-supported blades in which the main load-carrying member is the outside blade shell (refs. 3 and 4), (2) strut-supported blades in which the main load-carrying member is a strut that is submerged in the cooling medium (refs. 5 to 7), and (3) perforated blades in which an essentially solid airfoil is drilled, cored, or extruded to form spanwise coolant passages (ref. 8). Generally, of the three main blade types, the shell-supported blades are the lightest weight. The strut blades are best cooled but heavier and higher stressed, and the perforated blades are most easily fabricated. However, the latter are considerably heavier and require appreciably more cooling air.

It seems reasonable that a combination strut- and shell-supported blade might utilize the better characteristics of each type. A disadvantage of most shell-supported blades is the high braze strength required in the joint between the shell and the mechanical root attachment. The combination blade can overcome this disadvantage by brazing the shell to a short strut which provides ample braze area and results in low shear stresses in the braze material. The weight and higher base stresses of the strut-type blade can be reduced by terminating the strut where the outer shell needs no additional support. This report presents the stress analysis and engine evaluation of such a blade design.

EFFECT OF BLADE PROPORTIONS ON BLADE RUPTURE LIFE

Stress Distribution in Blade

The blade was designed to meet the following operational requirements: maximum tip speed, 1300 feet per second; tip diameter, 26 inches; and airfoil length, 4 inches. The blade construction is shown in figures 1 and 2. The outer shell is self-supporting to the point where centrifugal stresses in the shell approach the highest permissible level. From there toward the base, the loads in the shell are transferred to an internal strut which is completely bathed in cooling air. For cooling surface augmentation, corrugated sheet such as described in reference 4 is brazed between the shell and strut.

In designing the blade the first thing to be determined is the effect of strut length on the stress at the critical points in the blade. One critical point would be the shell stress at the end of the internal strut. The other would be in the strut at or near the base of the

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airfoil. An analysis was made fulfilling the above requirements, and the results are shown in figure 3. The centrifugal stresses at the two critical points are plotted against strut length from the airfoil base. The curve with the negative slope indicates the centrifugal stress in the shell material at the end of the strut. When the strut length is zero, the blade is shell-supported, and the stress at the base of the untapered shell is 53,300 pounds per square inch. The shell stress is independent of the thickness of the outside shell material, and the same curve very closely gives the stress distribution for the corrugations, the inner sheet-metal island, and any other untapered part of the blade regardless of the shape of the airfoil profile.

The family of curves with the positive slope in figure 3 determines the stress at the base of the strut for various shell thicknesses and strut lengths. In order to establish these curves a number of items had to be selected arbitrarily. First, the blade cross-sectional profile at various spanwise stations had to be chosen. For this case, to retain proper turbine-compressor matching, the base cross-sectional profile of a standard solid metal blade was used for the cooled-blade design. The tip cross section had to be increased in area to allow the cooling air to escape from the end of the shell. Straight-line elements were used between the tip and the base to ease fabrication of the cooling corrugations. The corrugations were designed with both an amplitude and pitch of 0.050 inch and a thickness of 0.005 inch. Calculations using reference 9 show this configuration to be very effective for blade cooling. The inner sheet-metal island only confines the cooling air close to the outer shell where it is needed. A thickness of 0.007 inch was judged adequate for this purpose. The effect of varying the thickness of the outer shell and varying the pitch and amplitude of the corrugations will be analyzed and discussed later in the report.

Determination of Optimum Strut Length

The best strut length cannot be determined from the stress distribution in figure 3 alone. The optimum length is highly dependent on the temperature distribution along the span of the blade. Actual measurements of temperature distribution had not been completed, and, therefore, in order to continue the analysis the temperature profile shown in figure 4 was assumed which is based on past experience with cooled blades. The ultimate goal of the blade designed herein is to operate at appreciably higher gas temperatures than used in conventional jet engines today and, under these conditions to cool the blades to bring the metal temperatures down to the same range as in the solid uncooled blades.

The variations of the strut and shell material rupture properties as obtained from the general literature for different temperatures are shown in figure 5. For the strut material (S-816), a band of data points was found, but the minimum strength properties were used for further analysis. By using these stress-rupture data, the assumed spanwise temperature

profile of figure 4, and the calculated stresses of figure 3, the stress-rupture lives of both the outer shell and the strut can be estimated as a function of strut length. The case for the 0.020-inch-thick shell is plotted in figure 6. The intersection of the two curves gives the optimum strut length and the maximum blade life. A shorter strut would produce failure in the outer shell in less time, and, conversely, a longer strut will result in strut failure also in less time. In fact, the strut length as it affects blade life is rather critical. One-half inch too long or too short will cut the blade life to one-half its maximum value.

Effect of Blade Temperature Profile

To illustrate further the importance of spanwise temperature distribution on maximum blade life and optimum strut length, additional analyses were carried out for several arbitrary temperature profiles. The profiles investigated are shown in figure 7. The originally assumed profile was raised and lowered 50° F. Next, it was shifted radially outward 1/2 inch on the blade. In the analysis the stress-rupture curves for the blade materials shown in figure 5 were again used. The effect of the temperature changes are presented in figure 8. The curves are for a constant shell thickness of 0.020 inch. In all cases the temperature at the strut base was assumed to be 1200° F, corresponding to the value measured in most conventional engines. Therefore, a single curve indicates the strut life. The following values were interpreted from figures 6 and 8:

Temperature profile	Maximum life, hr	Optimum strut length, in.
Assumed	750	1.84
Raised 50° F	420	2.19
Lowered 50° F	1540	1.43
Shifted out 1/2 in.	1440	1.47

In an air-cooled blade the temperature level of the blade can be varied appreciably by varying the amount and the distribution of the cooling air. From figure 8 and the previous table it can be seen that small changes in temperature level can have pronounced effects on blade life and optimum strut length. Raising the temperature 50° F reduced the life 40 percent while lowering it 50° F doubled the life. Shifting the whole temperature pattern radially outward on the blade had very close to the same effect as lowering it.

It is apparent from this temperature analysis that, in order to accurately design the best blade of the type described herein, first the spanwise temperature profile must be known for the actual amount of cooling airflow intended in the particular application.

Effect of Outer-Shell Thickness

The analysis was extended in a similar manner to determine the effect of shell thickness on blade life and optimum strut length (fig. 9). In this analysis the same outside blade profile, corrugation amplitude, pitch, and thickness as well as inner-island shell thickness were maintained for all outer-shell thicknesses. Increasing the outer-shell thickness thus resulted in decreasing the strut cross-sectional areas and increasing the load the strut must carry.

As mentioned earlier, shell stress is independent of shell thickness. Consequently, shell life also is independent of shell thickness for a given strut length. Therefore, only the single curve in figure 9 is needed to indicate the shell life for all shell thicknesses. Because more than one curve is required to show strut life and, thus, a different optimum strut length (indicated by the curve intersections) occurs for each shell thickness, a different maximum shell life and blade life result for each shell thickness. Decreasing the outer-shell thickness appreciably increased the maximum blade life and increased the optimum length of the strut.

The effect of shell thickness is shown better in figure 10 where the shell thickness is plotted against the maximum blade lives as established by the intersection of strut and shell life curves of figure 9. Halving the shell thickness from 0.020 to 0.010 inch would give almost three times the blade life (740 to 1950 hrs) for the assumed temperature conditions of this analysis. However, other important factors must be considered when reducing outer-shell thickness. The reduction is seriously limited by difficulties in welding leading and trailing edges and the lack of impact and erosion resistance. A thin shell, too, is more susceptible to vibration failure.

Effect of Corrugation Amplitude

Another variable investigated was the effect of corrugation amplitude for various shell thicknesses again while maintaining the blade outside dimensions constant. A series of graphs similar to figures 6 and 9 were computed for various corrugation amplitudes. Then the maximum blade lives were established for the optimum strut lengths, and these results are presented in figure 11. Obviously, the blade lives can be substantially increased by a reduction in corrugation amplitude. Unfortunately, there is a practical limit to the reduction in corrugation

amplitude dictated by the decrease in cooling efficiency, increased pressure drop, and the tendency of foreign matter to block the cooling passages.

Effect of Corrugation Pitch

An additional variable that was investigated was corrugation pitch for a fixed corrugation amplitude of 0.050 inch. The effect is shown in figure 12. An increase in pitch increases the maximum blade lives for both the 0.010- and 0.020-inch-thick shells in roughly the same proportions. This increase is due to the weight of the corrugations decreasing with increased pitch. Increasing the pitch, however, reduces the number of apexes to be brazed to both the shell and the strut, thereby increasing the shear stresses in the braze as well as reducing the heat transfer (ref. 9). Because of the stringent requirements and complexity of brazing, 100-percent effective shear area cannot be guaranteed, and, therefore, an excess in theoretical shear area must be provided. (For all optimum strut lengths, more than adequate theoretical shear area is provided.) A thinner shell can safely utilize a larger corrugation pitch than the thicker shell because the centrifugal shell load and, consequently, the shear stress are directly proportional to shell thickness.

Proportions Selected and Resulting Stresses for Test Blades

A thickness of 0.020 inch was selected for the outer shell although a 0.010-inch-thick shell appears very promising. Skin erosion, shell vibrations, resistance to impact by foreign particles, and welding of the leading and trailing edges all favored the selection of the 0.020-inch-thick shell. By considering the results of figure 6, a strut length of 1.792 inch was chosen. To provide the maximum practical amount of augmented cooling surface for higher temperatures (ref. 9) and to provide adequate braze shear area for the heavier shell, the corrugation amplitude and pitch were made to the minimum practical value, 0.050 inch each.

The stress distribution along the blade span was calculated for these proportions and is presented in figure 13. Also, the cross-sectional areas used in the computations along the blade span are included in the figure. At the end of the strut the area was considered equal to the sum of the areas of the shell, the corrugations, and the strut. From the end of the strut toward the blade base, the area was assumed to diminish linearly until at the base only the area of the strut was considered as carrying the whole blade.

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A large number of the cooled blades previously evaluated at the NACA appear to have failed because of blade vibrations. Particular attention was paid to this factor in the design of the blade strut. The strut surface was kept smooth without any fins, grooves, or sharp fillets which introduce localized stress concentrations in alternate bending. A very generous radius was provided at the base of the strut where the airfoil shape was blended into the mechanical root fastening.

The variation in stress-rupture life along the span of the blade for the previously assumed spanwise temperature profile (fig. 4) is presented in figure 14. This figure shows the two points along the span where the blade is likely to fail in stress-rupture with time. The actual times to failure are not important because they can be readily shifted up or down by slight changes in the amount of cooling air used.

BLADE MATERIALS AND FABRICATION

Because the blade is intended for eventual increased operating temperatures, the most suitable commercially available heat-resistant materials were used. All the sheet materials which included the outer shell, the corrugations, and the inner shell beyond the strut were made from Haynes Alloy No. 25 (L-605). The struts were machined from standard S-816 forged turbine blades. The parts were bonded together (in a vacuum furnace) with a high-temperature braze composed of 20 percent chromium, 10 percent silicon, 1 percent iron, and the balance nickel.

The outer shells (0.020-in. thick) were formed in halves, parting at the leading and trailing edges, by the stretch-forming process to avoid spring-back common with ordinary press-forming methods. The corrugations were pressed one corrugation at a time. On a production basis this material, which is only 0.005-inch thick, can be easily roll-formed. The inner-shell pieces are cut also in halves from 0.007-inch-thick stock.

In assembly, the three pieces forming either the pressure or suction surfaces were tack-welded together at the top end only. Collodion cement was used to help temporarily hold the inner-shell half to the corrugation. No prior forming was done to the inner shell. It was merely forced to conform to the contour of the outer shell. Braze material in the form of plastic bonded wire was threaded into each of the corrugation passages. The three pieces were then clamped in dies and placed in the vacuum furnace for brazing. A soft fibrous refractory padding material was used between the dies and the sheet-metal parts to assure a uniform pressure while brazing.

When finished, the airfoil halves were brazed to the strut, again using plastic braze wire, refractory padding, and clamped dies. The

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leading and trailing edges were heliarc-welded without filler rod. The
sheet-metal cover comprising the platform and the air seal for the turbine
rotor (fig. 1) was added, and the blade was subjected to the third braz-
ing cycle. The brazing cycles consisted of heating to 1800° F and hold-
ing for 10 minutes, followed by rapid heating to 2130° F in a vacuum of
1 to 4 microns of mercury and holding for 15 minutes. The blades were
then rapidly cooled in air. Finally, the blade tips were trimmed to
size which removes completely the tack-weld points.

ENGINE INSTALLATION AND PROCEDURE

The cooled blade roots were machined to slide into the serrations
of a standard rotor. Two cooled blades were mounted diametrically op-
posed in the rotor, and the ends of the sheet-metal air seal (fig. 15)
were bent-over and welded to the fore and aft faces of the rotor. The
two adjacent standard solid blades were grooved slightly under their
platforms so as to slide over and tightly seal the sheet-metal platform
of each cooled blade. Cooling air was brought to the blades through two
1/2-inch-diameter tubes welded to the rear face of the rotor. The means
of transferring air to the revolving rotor can be seen in figure 16.

Air from the laboratory air supply system was ducted to the engine
tailcone at pressures up to 90 inches of mercury. The amount of cooling
airflow into the engine was measured with orifice plates in supply lines.
A large part of this air, however, leaked by the labyrinth seal and
through the ball bearing of the air transfer device (fig. 16). This
leakage air was approximately measured by operating the engine at the
same conditions with the cooling tubes to the blades blocked.

The primary purpose of the engine running was to determine the en-
durance qualities of the cooled blades. Therefore, no blade temperature
or pressure loss measurements were taken on the engine. On the initial
engine run the speed was raised from idling speed by small increments
until rated speed was attained, this operation requiring about 1/2 hour.
Subsequent accelerations were normal (5 to 15 sec) for this engine. To
determine if mechanisms other than stress-rupture would produce blade
failure, relatively large airflows (3.5 to 4.0 percent of the mass flow
handled by each blade in the turbine) were used to cool the blades. The
engine was run continuously at a rated speed of 11,500 rpm and a turbine-
inlet gas temperature of 1660° F until the end of the shift or malfunc-
tion of other parts of the engine caused shutdown. After accumulating
100 hours of this type of operation, the blades were subjected to an
additional 25 hours in the engine at the same operating conditions with
the cooling air closed off to evaluate the stress-rupture strength of
the blades. The temperature profile for the conventional solid uncooled
blades has been measured for this engine and is presented in figure 17.

The profile for the cooled-type blade with all cooling air shut off should be approximately the same. Using this temperature profile, the stress-rupture life along the blade span was again determined and is shown in figure 18. From this figure it is obvious that the critical point is at the end of the strut and that stress-rupture failure, when uncooled, can be expected in a little over 100 hours.

The blades were removed from the engine and were checked in a bench test for cooling airflow distribution and pressure drop. New blades of identical design were also checked for flow characteristics for comparison purposes.

RESULTS OF ENGINE TESTING

After approximately 35 hours of engine running, cracks were noted in the fillet of the sheet-metal platform where the metal had been subjected to appreciable plastic deformation in the drawing process. These cracks did not grow in length with continued operation even though no attempt was made to repair or stop them. Figure 19 is a photograph of the more serious of the two cracks after 110 hours of running at maximum engine speed.

After 110 hours of operation, the bent-over air sealing strip on the front face of the turbine rotor began to rub on the stationary gas baffle. The rubbing completely cut through the bent-over part of the sheet-metal platform of one of the cooled blades. The inadequately supported bent-over segment tore off and, in passing through the turbine, damaged the tips of many of the blades. The damage to the cooled blades, however, did not appear to be more extensive than that to the solid blades (fig. 20). A new piece was welded to the sheet-metal platform and front rotor face, and the running was continued.

The cooled blades were still in serviceable condition at the end of the complete running time (fig. 21) although there was some evidence of surface deformation of the outer shell corresponding to the point of termination of the inner strut. This evidence would tend to confirm the results of the stress computations. In accordance with figure 18 the test blade when uncooled would be expected to fail in 100 hours at the point where the surface distortion was noted. The deformation was so slight that it did not show on a photograph.

The bench-test check of the airflow showed uniform distribution along the chordal periphery at the blade tip. The comparison between the new and the engine-tested blades (after 125 hr) showed very similar flow characteristics, indicating very little if any blockage of cooling air passages.

SUMMARY OF RESULTS

A design analysis was made for a new-style cooled turbine blade consisting of an outer sheet-metal shell and an inner load-carrying strut which spans only part of the airfoil length. Full-length corrugated cooling surfaces were interposed between the shell and the strut, and the unit was assembled by brazing. The proportions of the blade components were selected in accordance with the results of the computations. These blades were subjected to endurance-type operation at rated speed in a full-scale engine. The results of the analysis and the engine testing are as follows:

1. For the assumed spanwise temperature profile which was based on experience with cooled blades:

    (a) The maximum blade rupture life was highly dependent on the length of the internal strut.

    (b) Variations in outer-shell thicknesses greatly affected the maximum blade life. Decreasing the shell thickness from 0.020 to 0.010 inch theoretically increased blade life almost threefold.

    (c) Decreasing the amplitude of the corrugated finning pronouncedly increased the maximum rupture life for blades of this design because the strut area was increased and the corrugation weight was reduced.

    (d) Increasing the pitch of the corrugation also appreciably increased blade life theoretically because of reduced corrugation weight.

2. The optimum strut length to give the maximum blade life was highly dependent on the temperature gradient along the blade length.

3. Blades designed in accordance with the analysis endured 125 hours at rated speed and temperature in the engine without failure, including 25 hours without cooling airflow through the blades.

Lewis Flight Propulsion Laboratory
    National Advisory Committee for Aeronautics
        Cleveland, Ohio, May 3, 1957

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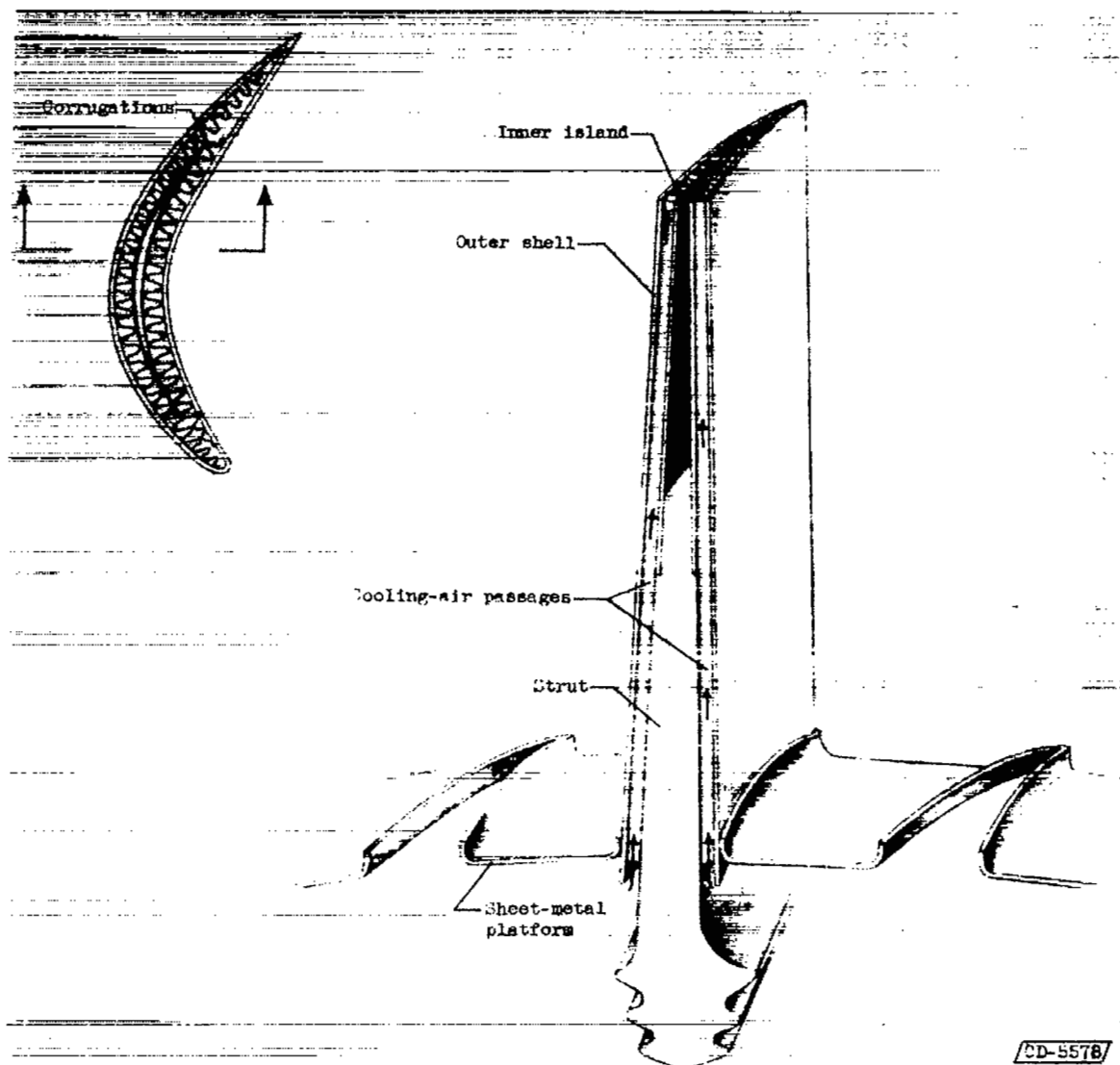
Figure 1. - Cross sections of cooled turbine blade showing construction.
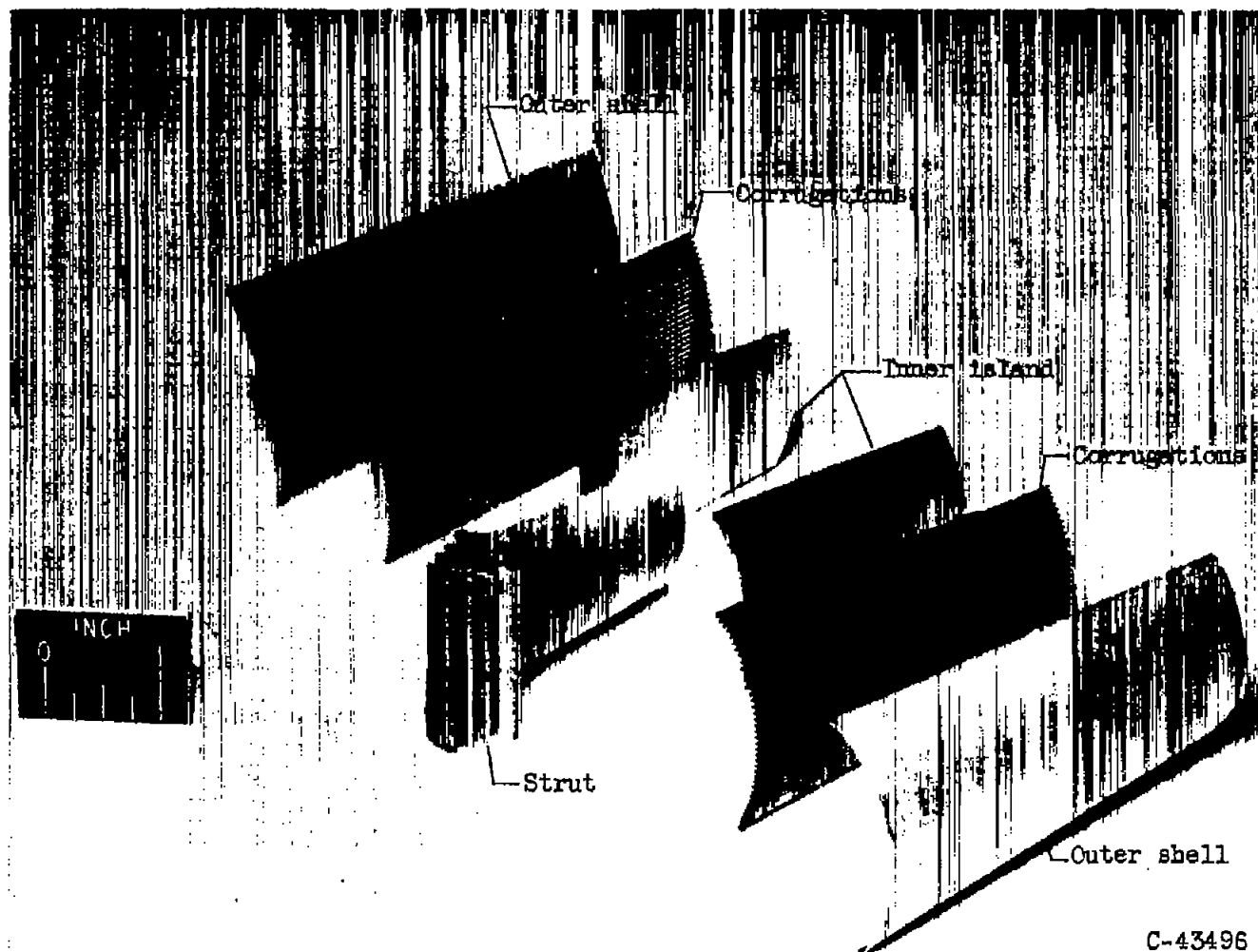
Figure 2. - Exploded view of blade components.

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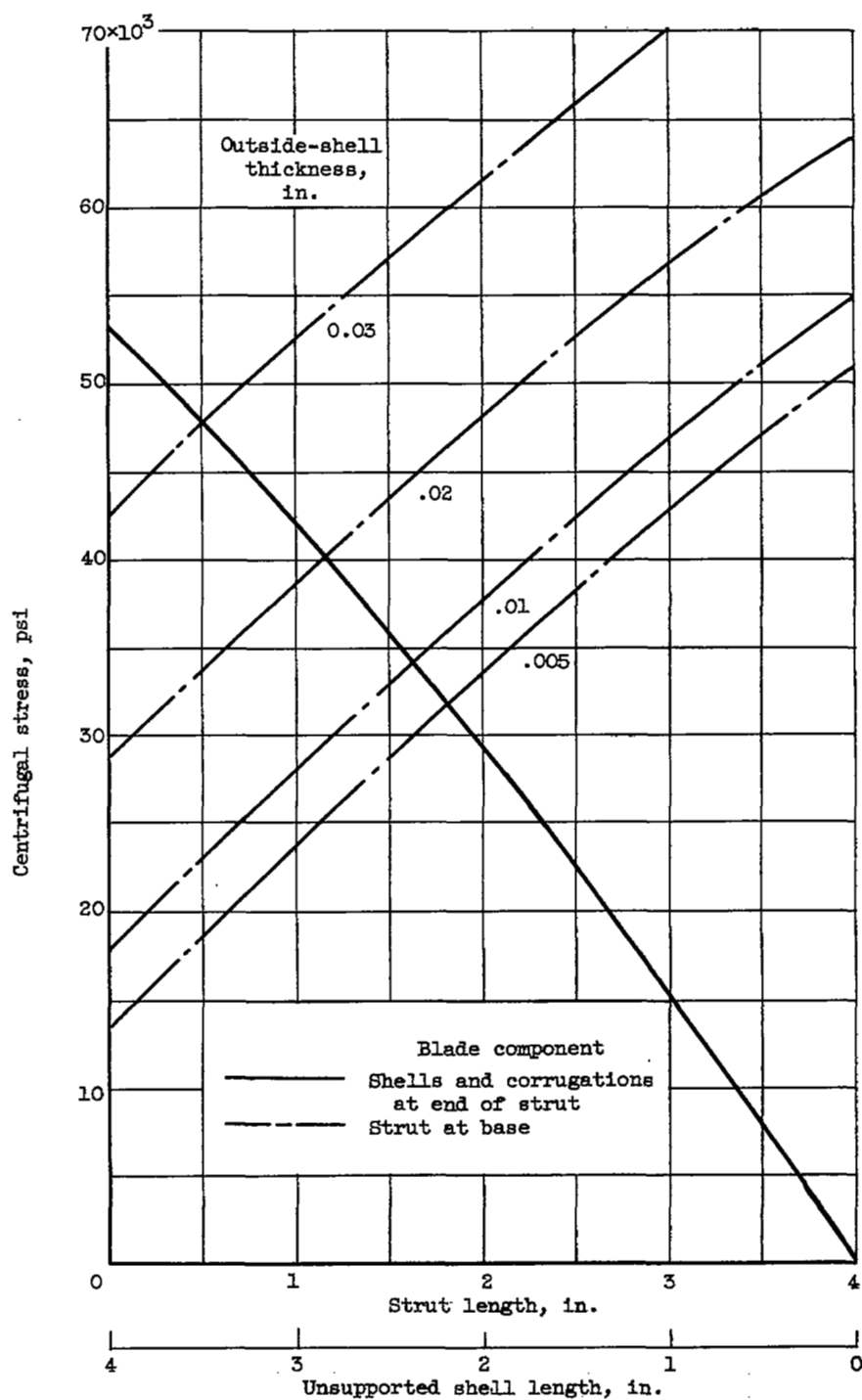
Figure 3. - Stresses at critical points of blade for various
shell thicknesses and strut lengths.
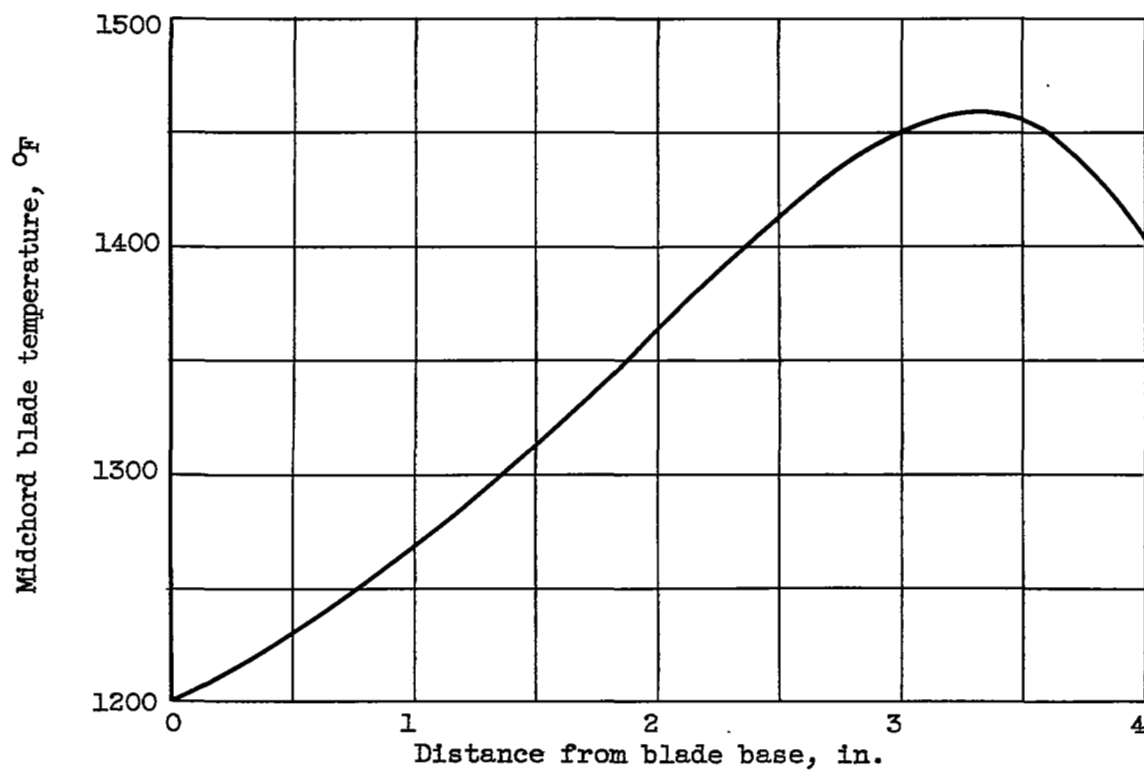
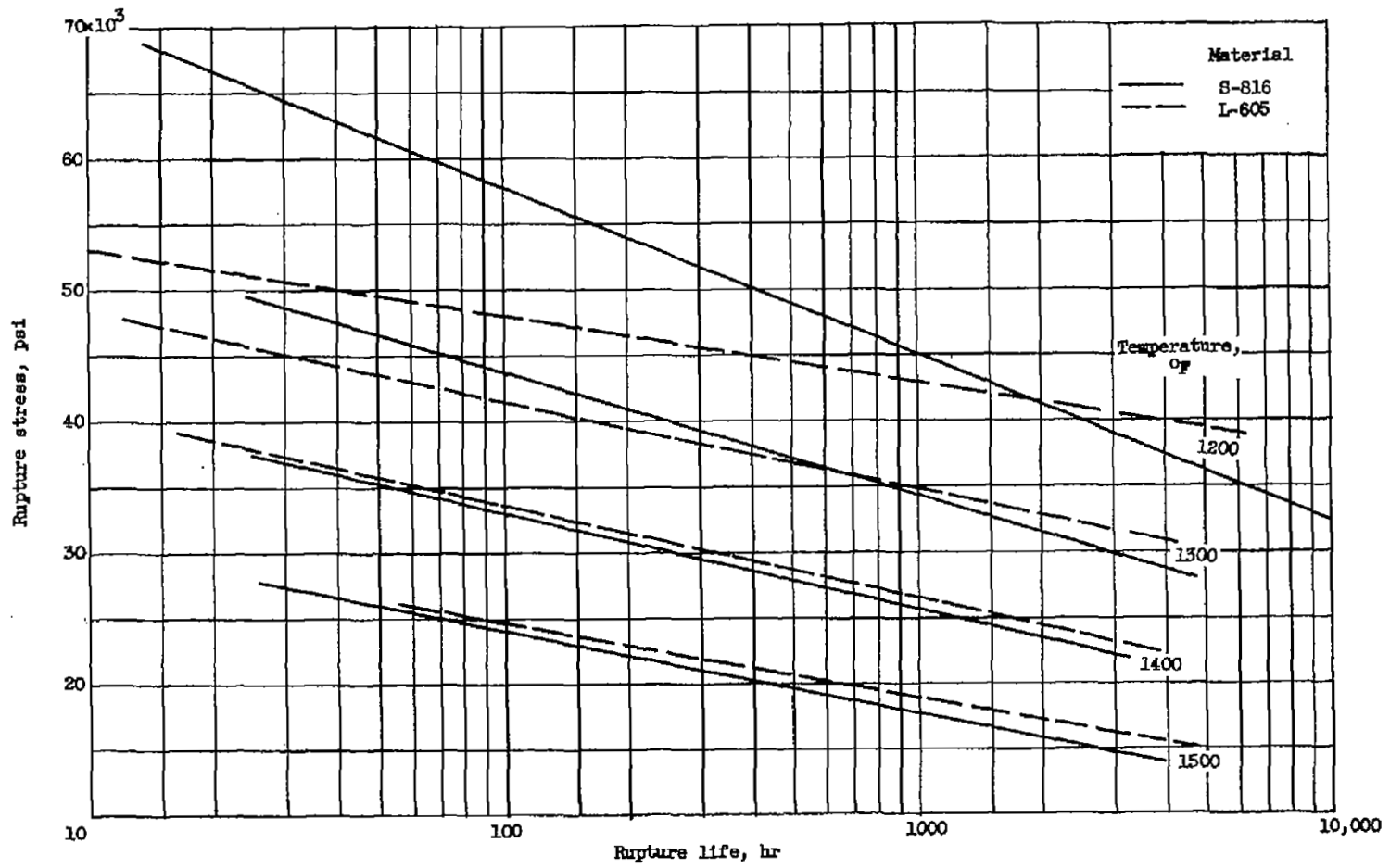Figure 4. - Assumed spanwise temperature profile for cooled
blade.

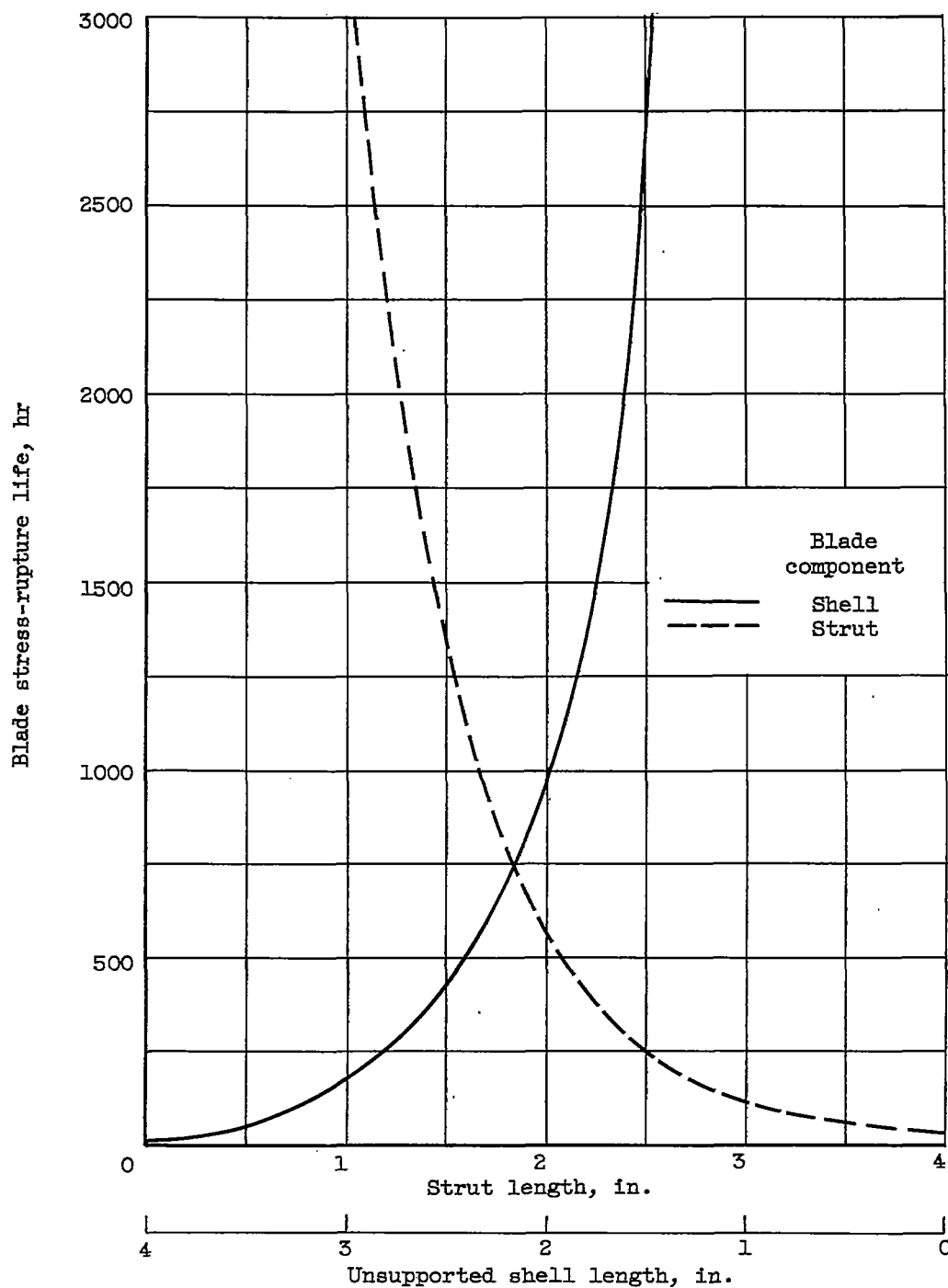Figure 5. - Minimum stress-rupture properties for cooled blade materials.

Figure 6. - Determination of optimum strut length and maximum
blade stress-rupture life for assumed temperature profile and
0.02-inch-thick outer shell.
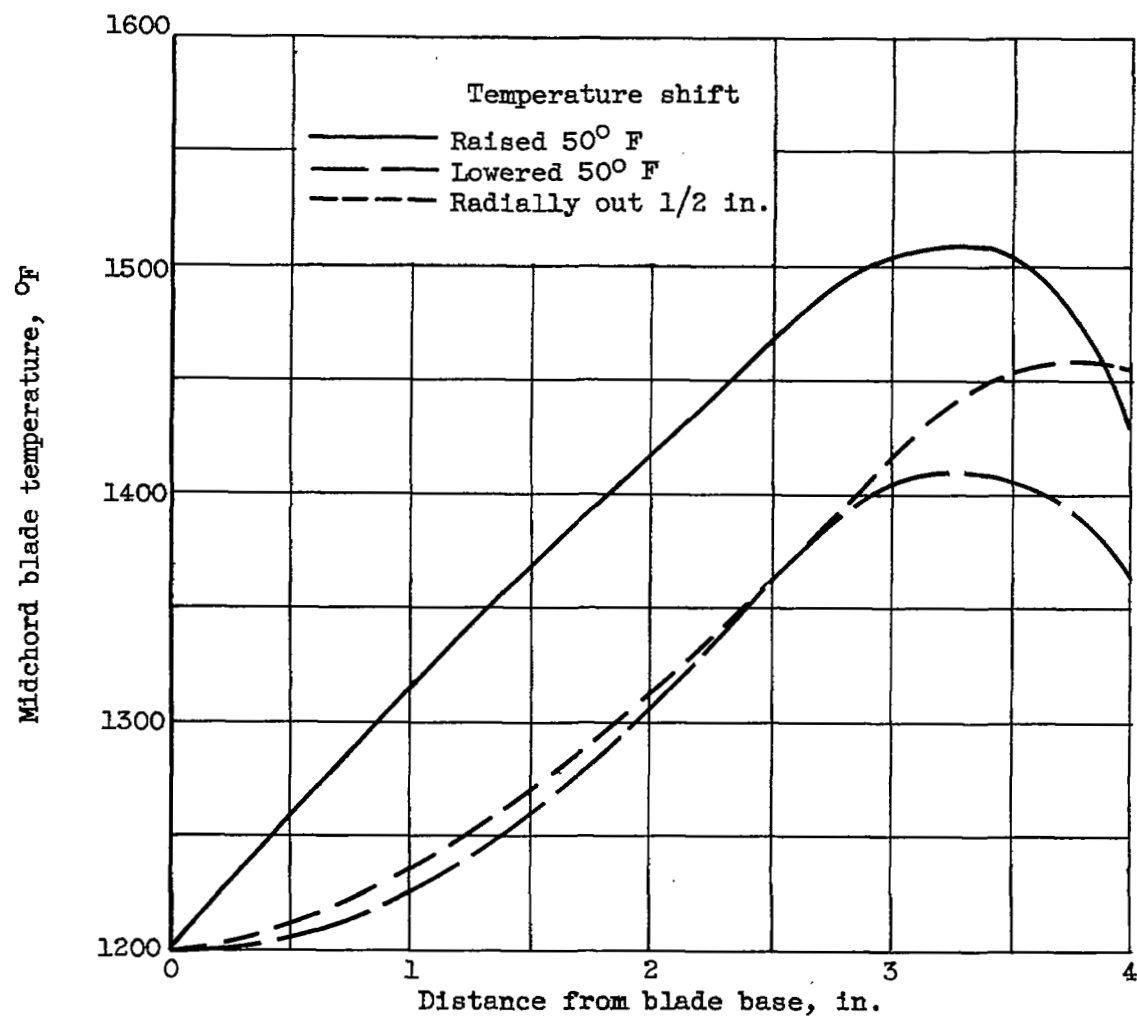
Figure 7. - Shifts in assumed temperature distribution.

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Blade stress-rupture life, hr

Strut life

Shell life for various
temperature shifts

———————— Raised 50° F
— — — Lowered 50° F
– – – – Radially out 1/2 in.

Strut length, in.

Unsupported shell length, in.
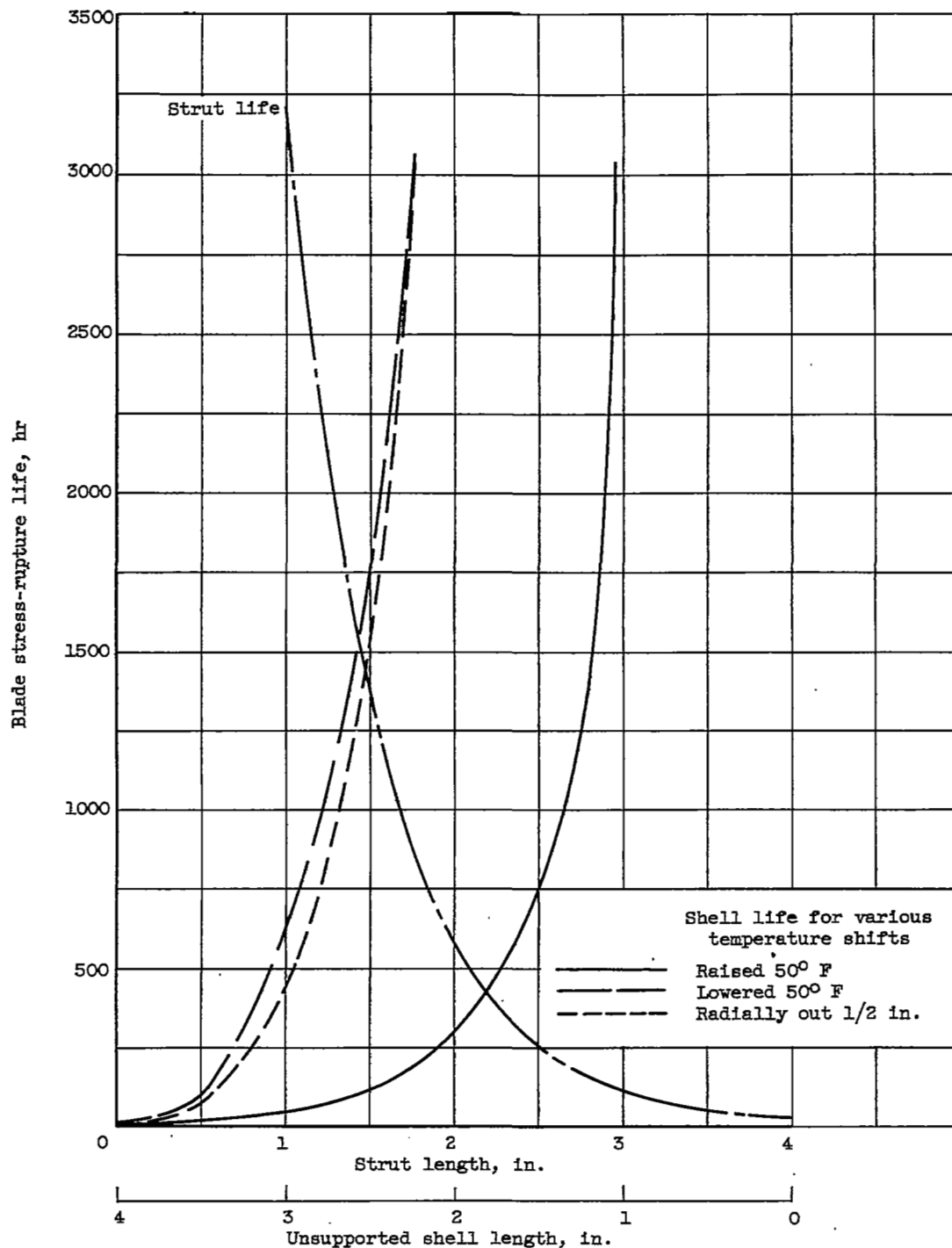
Figure 8. - Effect of temperature shifts on blade stress-rupture life and
optimum strut length. Outer shell, 0.02-inch thick.
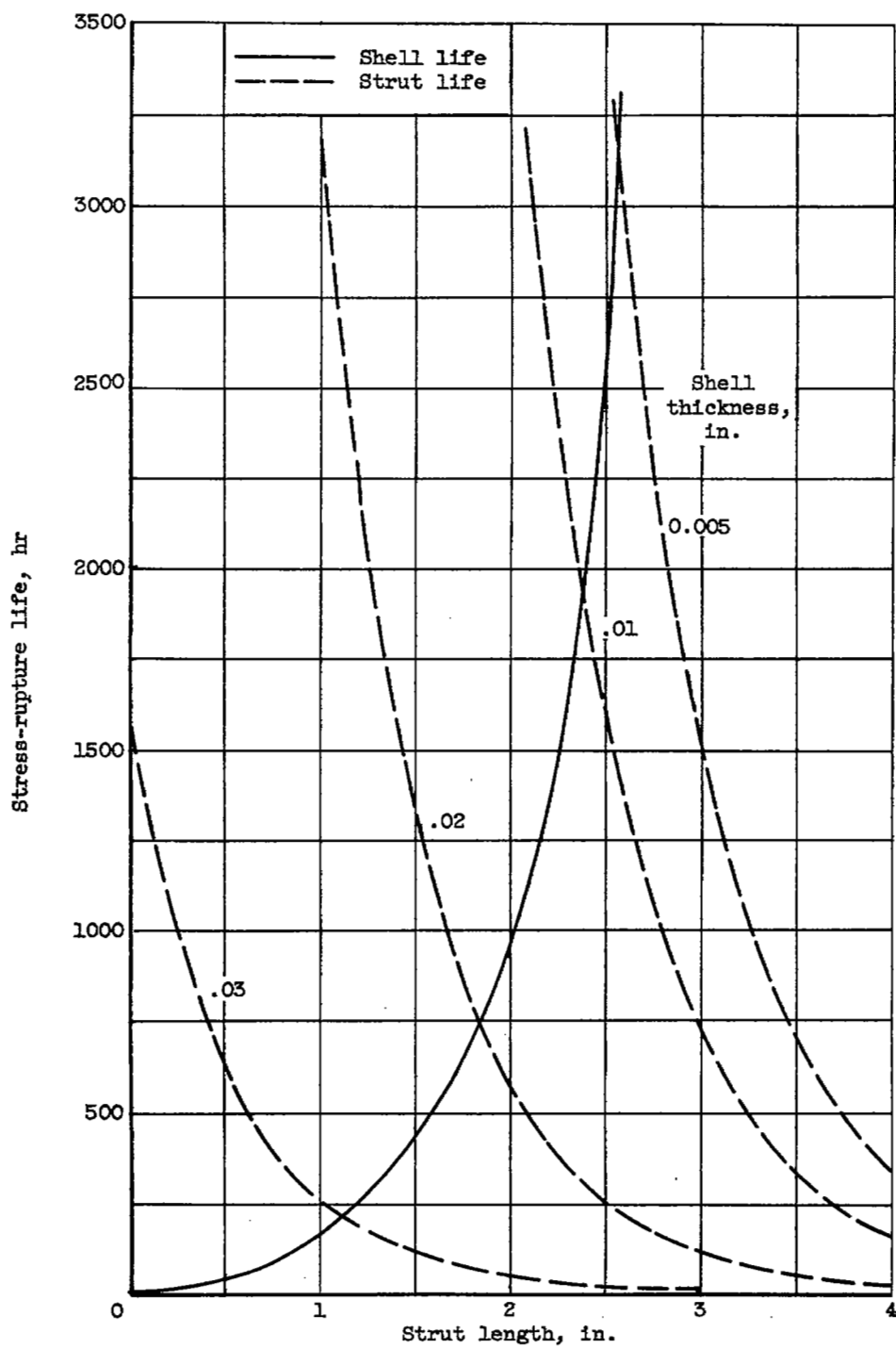
Figure 9. - Effect of outer-shell thickness on optimum strut length for assumed temperature distribution.

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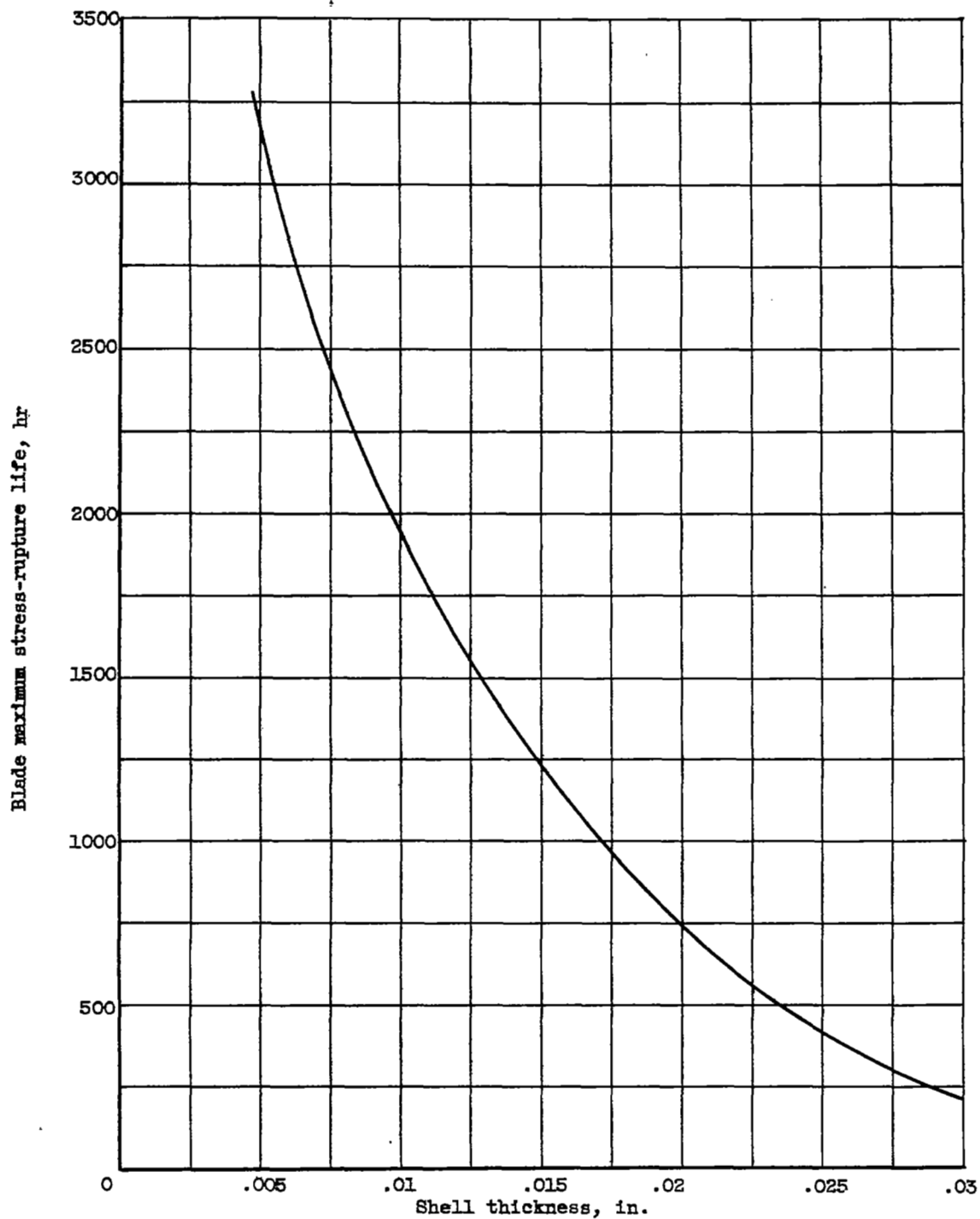
Figure 10. - Effect of outer-shell thickness on maximum blade stress-rupture life for assumed temperature distribution.
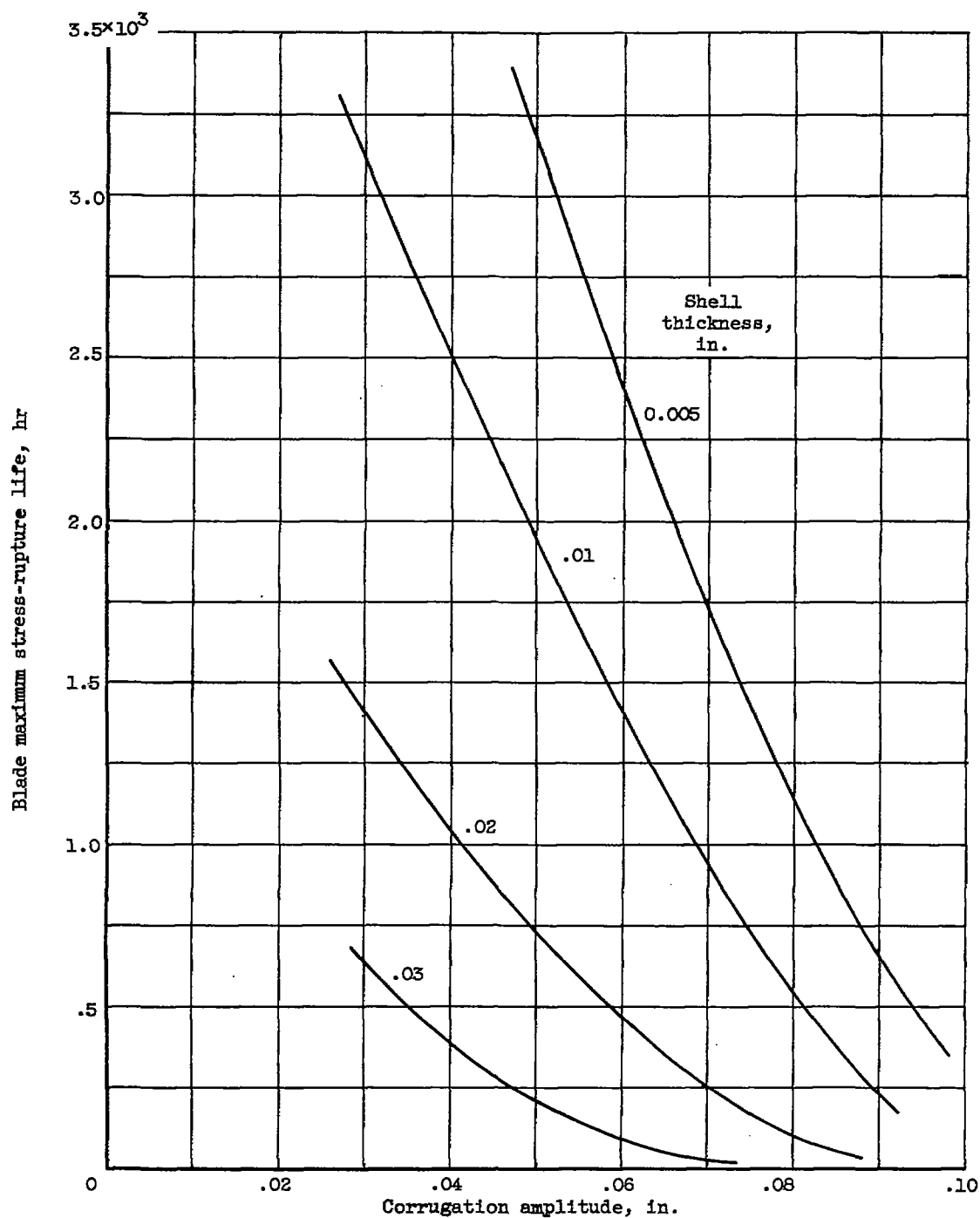
Figure 11. - Effect of amplitude of cooling fin corrugations on blade stress-rupture life for various shell thicknesses. Corrugation pitch constant, 0.05 inch.

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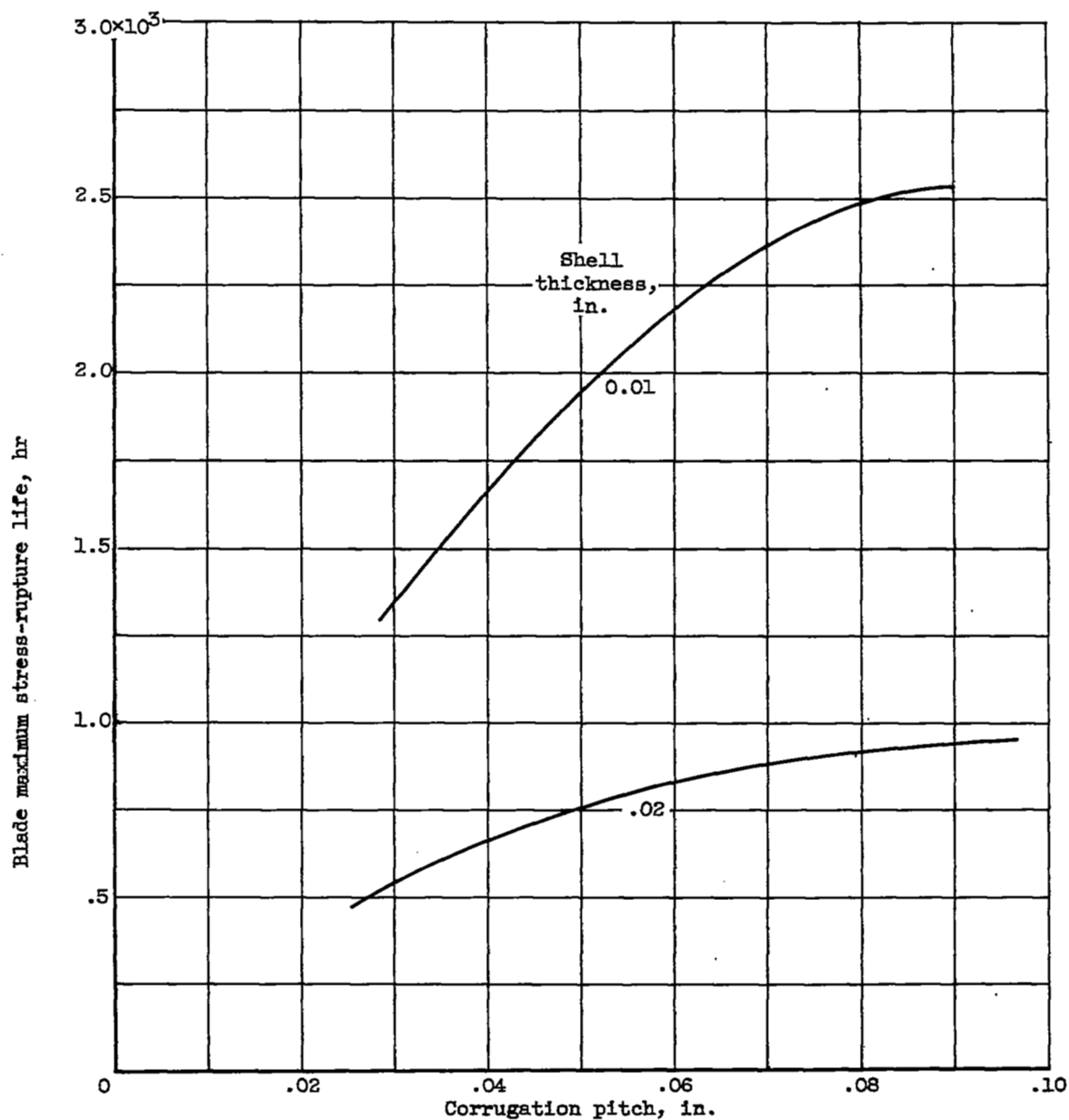
Figure 12. - Effect of pitch of corrugations on blade stress-rupture life for various shell thicknesses. Corrugation amplitude constant, 0.05 inch.
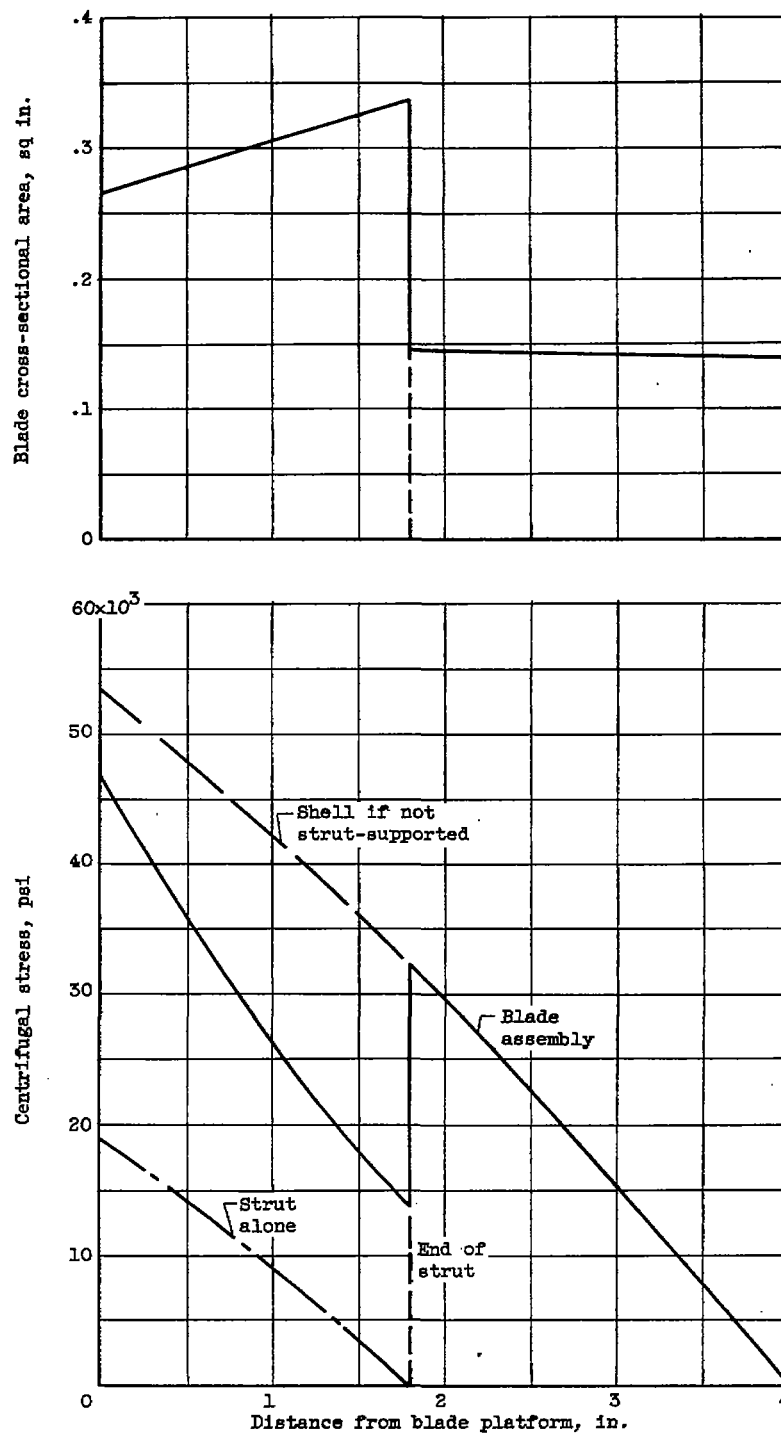
Figure 13. - Cross-sectional area and stress distribution along cooled blade span.

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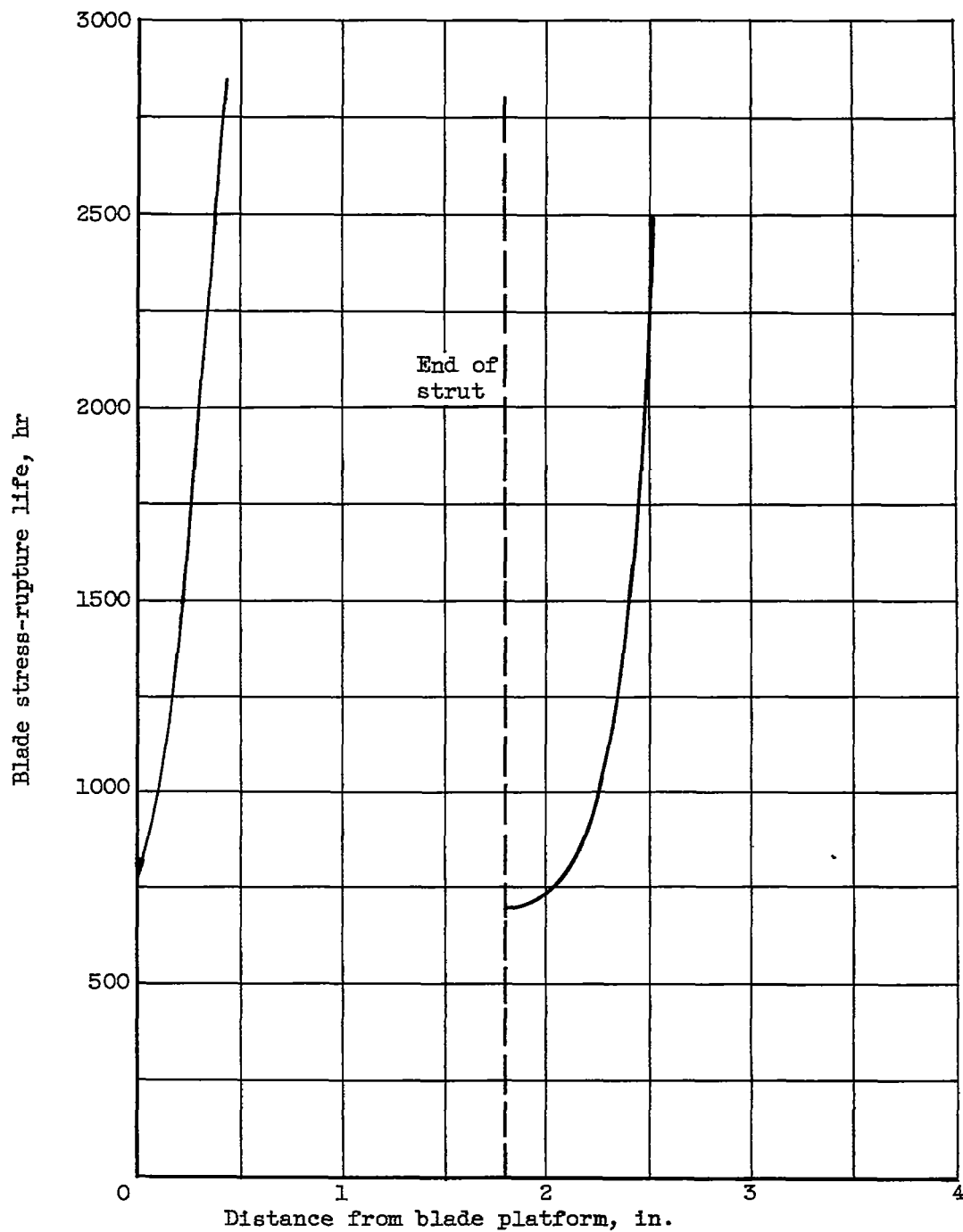
Figure 14. - Stress-rupture life along blade span for assumed temperature distribution.
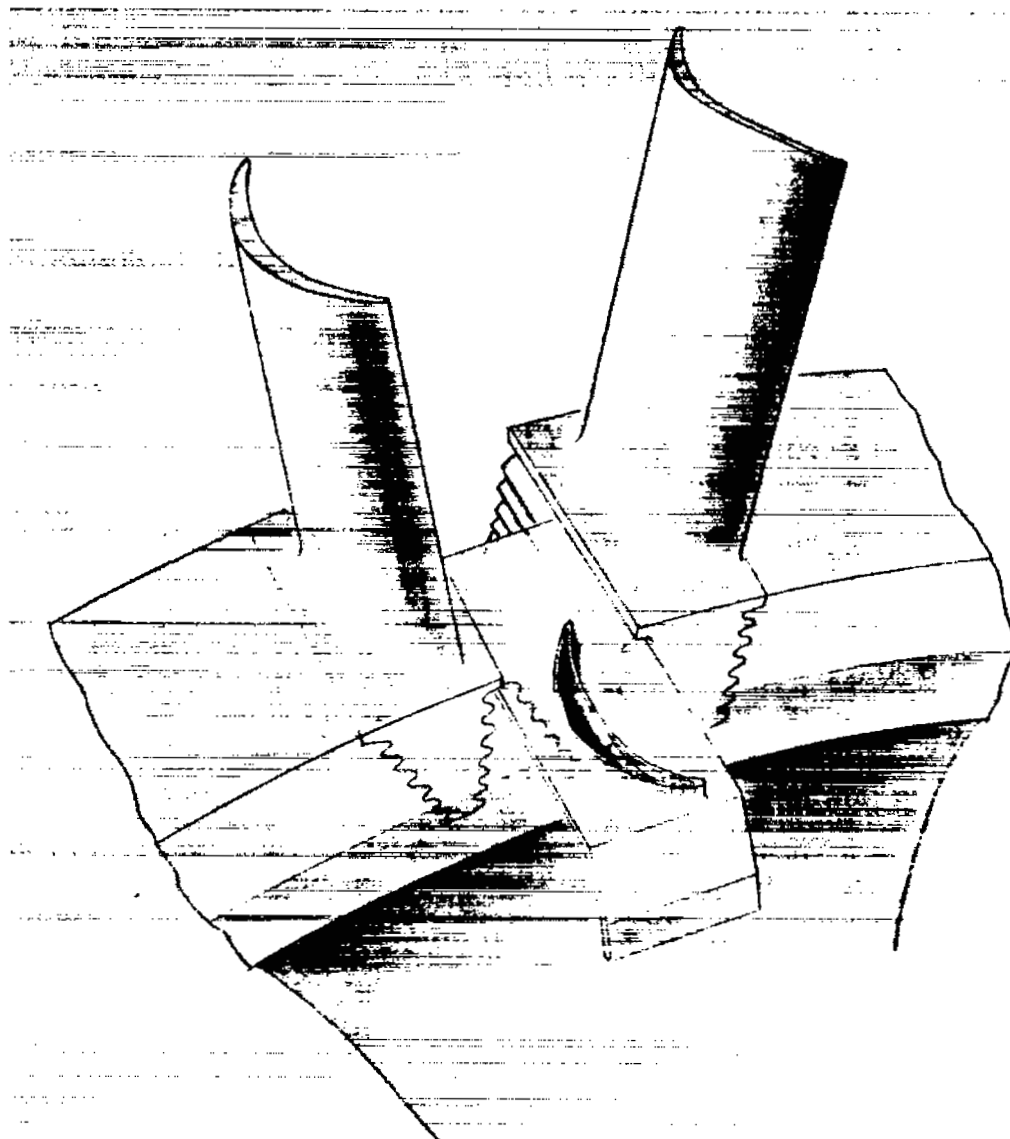
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Figure 15. - Sheet-metal rim cap.

Cooled turbine
blade

Air transfer
tubes

Labyrinth
seal

CD-5526

→ Blade cooling air

⇐ Tailcone cooling air

⇐ Oil mist for bearing
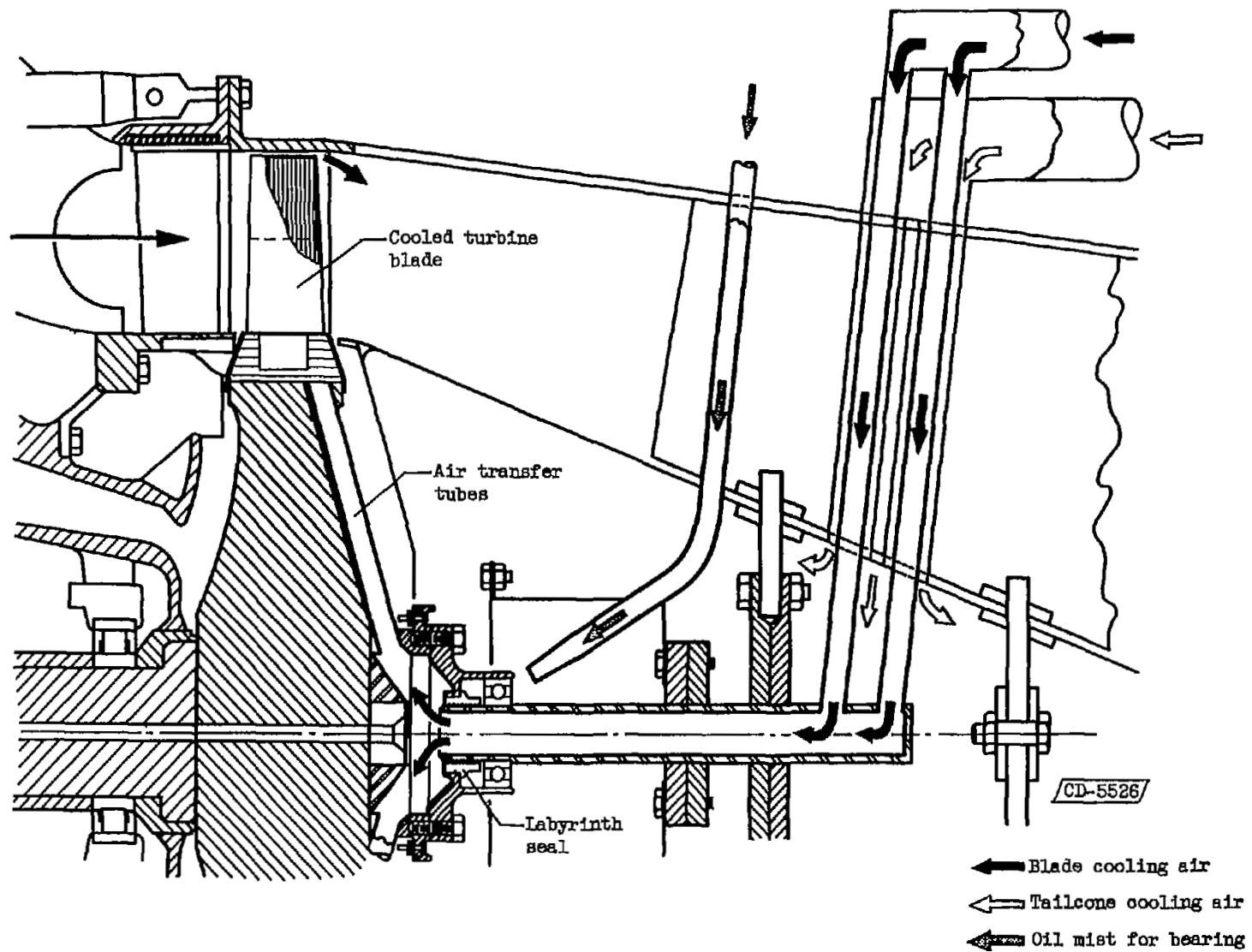
Figure 16. - Cooling air transfer device for engine tests with two cooled blades.
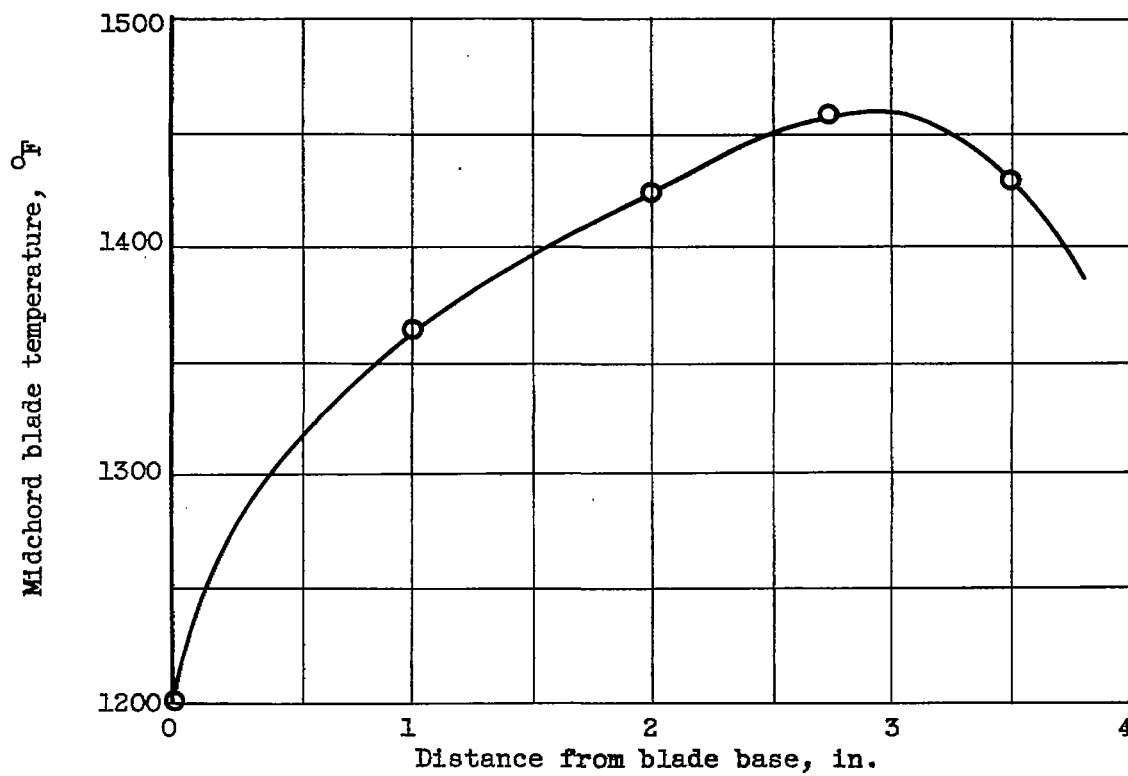
Figure 17. - Temperature profile for uncooled blades under maximum engine operating conditions.

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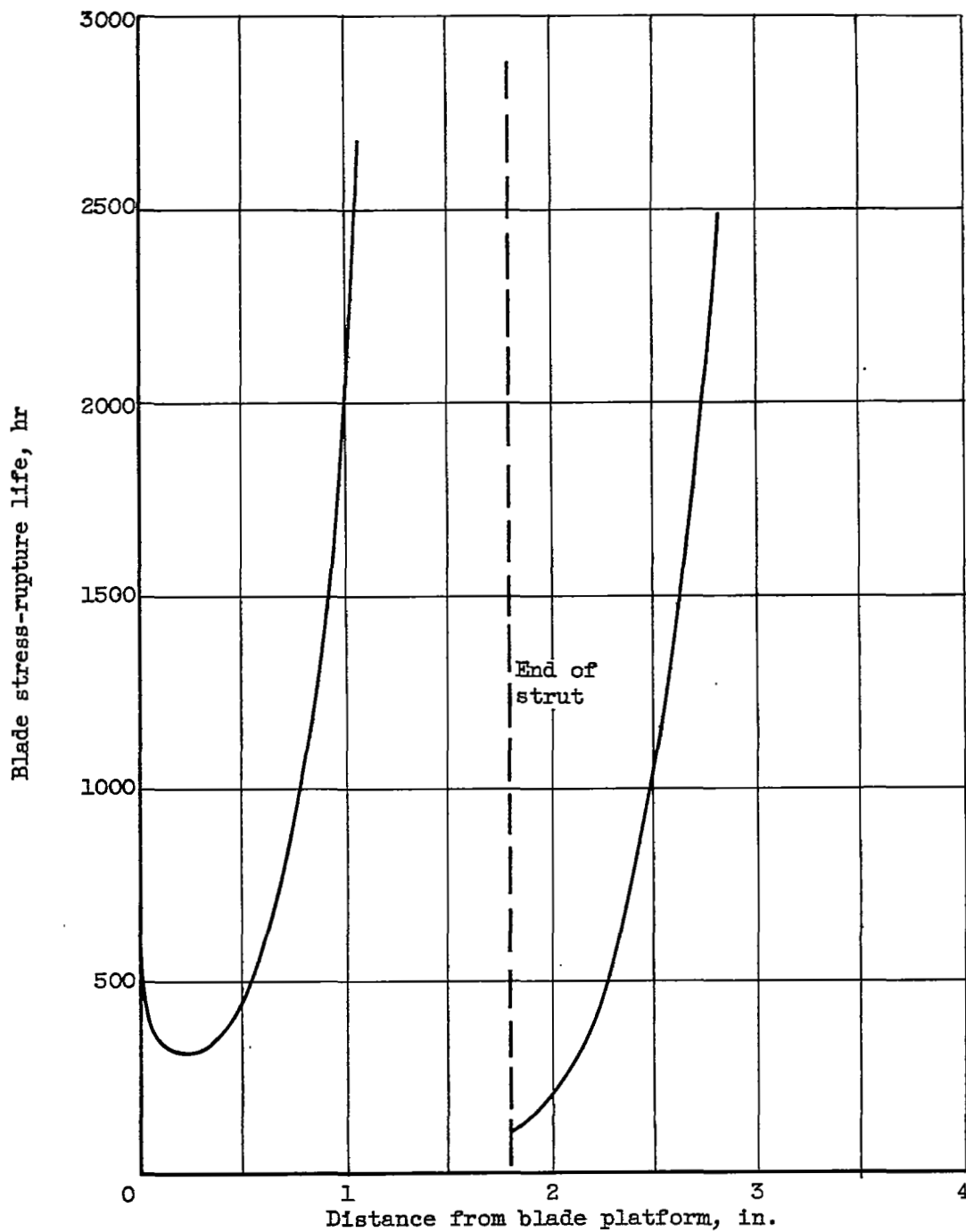
Figure 18. - Stress-rupture life along blade span when cooling
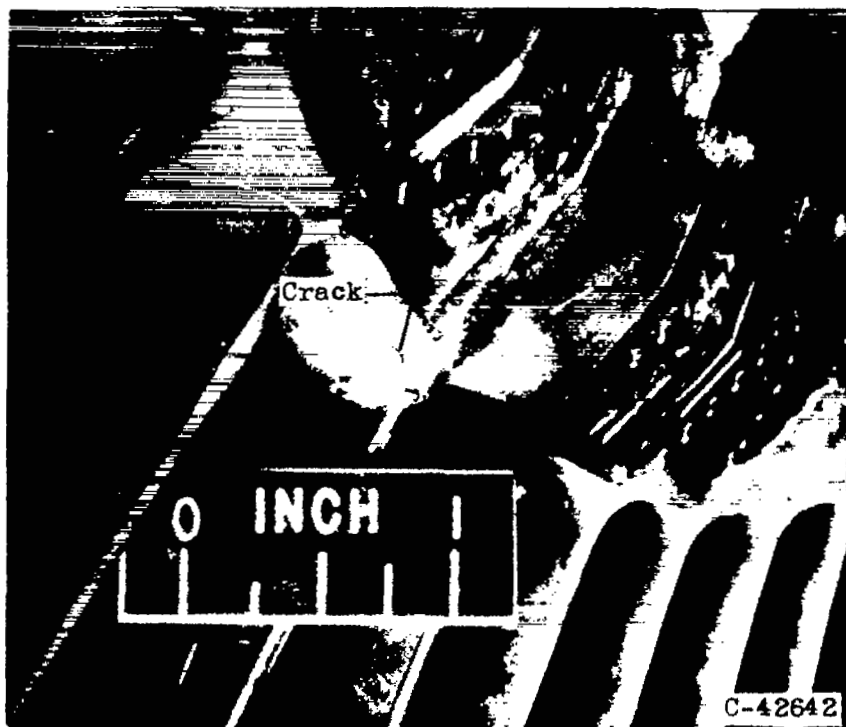air is shut off from test blade.

Figure 19. - Crack in sheet-metal platform after operation
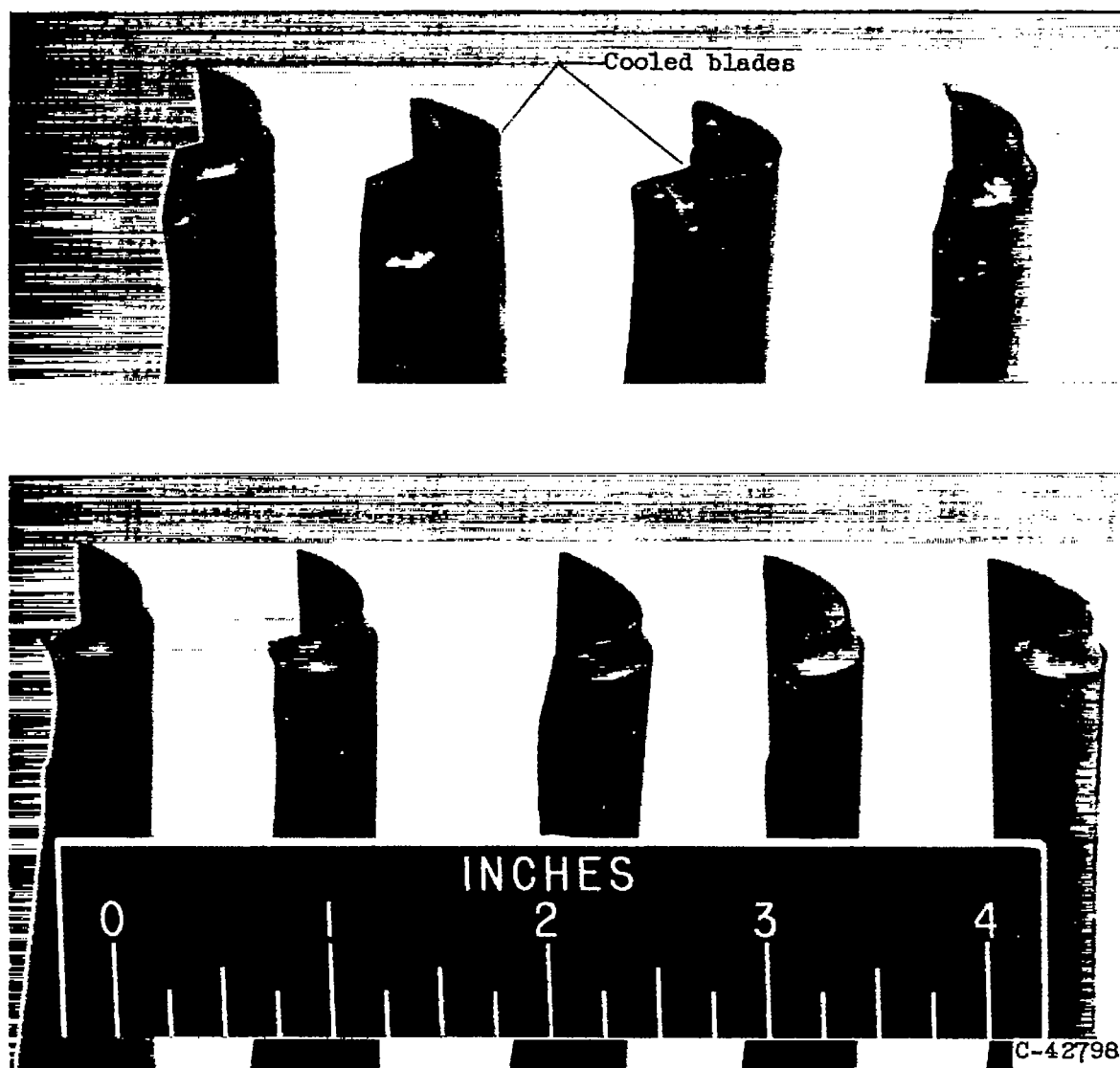for 110 hours at rated speed.

▬▬▬▬

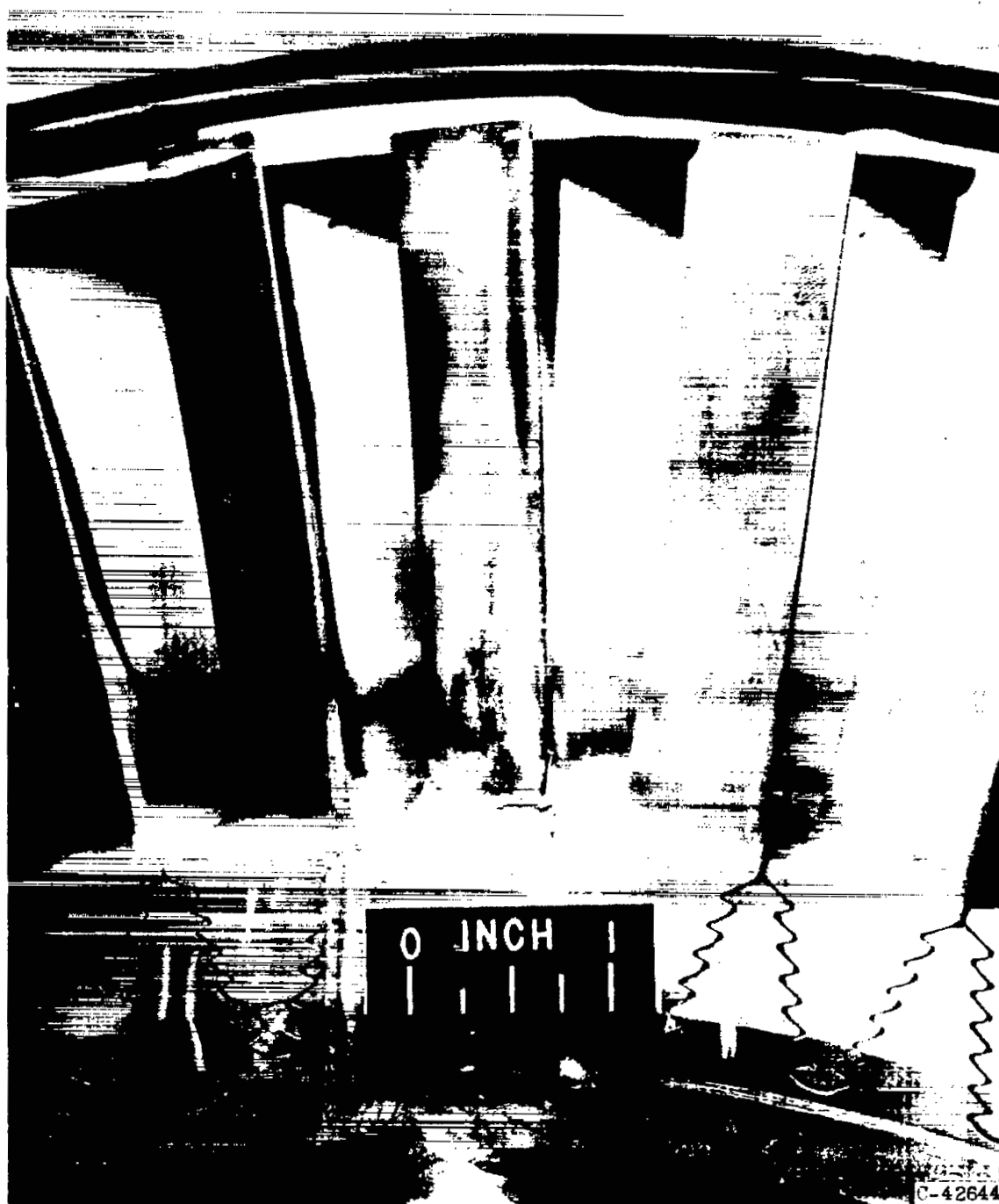Figure 20. - Damage to blade tips caused by piece of rim cap.

Figure 21. - Blade appearance after 125-hour engine run.